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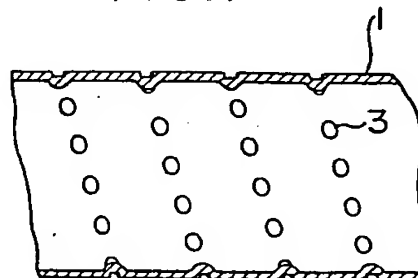
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(54) Heat transfer tube for single phase flow.

(57) A heat transfer tube for a single-phase flow having rows of discontinuous projections (3) formed on the inner surface (1) thereof along one or more spiral curves (4). Each projection (3) has a circular (32), elliptic (34) or a similar cross-section (36, 38) constituted by smooth curves at any desired height including the bottom thereof. The cross-sectional area of the projection (3) progressively decreases towards the top of the projection (3).

FIG. 1



HEAT TRANSFER TUBE FOR SINGLE PHASE FLOW

The invention relates to a heat transfer tube for single-phase flow having one or more rows of projections formed on the inner surface of said tube along a spiral curve or curves, each row having a multiplicity of projections formed discontinuously, the portions of the tube inner surface between adjacent rows presenting surfaces parallel to the tube axis, as well as to a method for producing such a tube.

Such heat transfer tubes are used in heat exchangers of, for example, air conditioners, refrigerators and so forth, and are suited particularly to heat transfer between a single phase flow in the tube and a fluid flowing outside the tube.

As well known to those skilled in the art, heat exchangers of air conditioners and refrigerators incorporate heat transfer tubes. Various types of heat transfer tubes have been proposed up to now. Some of these heat transfer tubes have smooth inner surfaces, while other heat transfer tubes have two- or three-dimensionally machined surfaces. For instance, the specification of United States Patent No. 3,768,291 shows a heat transfer tube having two-dimensional ribs formed on the inner surface thereof. On the other hand, the specification of United States Patent No. 3,830,087 discloses a heat transfer tube in which a rolling plug is driven into the tube blank such as to effect a grooving thereby forming primary ribs and then an additional machining is conducted to form secondary grooves, thus providing the tube inner surface with three-dimensional projections.

1 The heat transfer tubes having two- or three-
dimensionally machined inner surfaces encounter the following
problems when used for a single phase flow of a fluid.
Namely, since the edges of the projections on the tube inner
5 surface are not rounded by are sharp, exfoliation eddy cur-
rent are formed in the fluid when the fluid turns around
the sharp corners or edges, so that a large pressure drop is
caused between the inlet and outlet ends of the heat
transfer tube, requiring a greater power for driving the
10 fluid through the tube. In addition, the fluid tends to
stagnate on the rib surfaces perpendicular to the streamline
so that the kinetic energy possessed by the fluid is changed
into collision pressure to cause a wear of the ribs during
a long uses. As a result, the heights and shapes of the
15 ribs are gradually changed from the optimum design heights
and shapes, resulting in a degradation of the heat transfer
performance.

In addition, the method for forming ribs by rolling
plug essentially requires a troublesome work including the
20 primary grooving and secondary grooving, resulting in a
raised production cost of the heat transfer tube.

Accordingly, the object of the invention is to
provide a heat transfer tube for single phase flow, having a
high heat transfer rate and provided with a highly durable
25 construction of the heat transfer surface, as well as a
method which permits the production of such a heat transfer
tube at low cost.

To this end, according to the invention, there is provided a heat transfer tube as mentioned at the beginning, wherein each of said projection has a circular or elliptic or a similar cross-section constituted by smooth curves at its any portion along the height including the bottom thereof, the cross-sectional area of said projection progressively decreases towards the top thereof.

Advantageously, said projections have a height of 0,45 to 0,6 mm and are arranged at a circumferential pitch of 3,5 to 5 mm. They may have an axial pitch of 5 to 15 mm.

Preferably, porous heat transfer surfaces are formed on the outer surface of said tube.

The method of producing a heat transfer tube for single-phase flow having one or more rows of projections formed by a plastic work on the tube inner surface along a spiral curve or curves, each row including a multiplicity of discontinuous projections such that the portions of the tube inner surface between adjacent rows presenting surfaces parallel to the tube axis comprises the steps of depressing the outer surface of a tube blank by a tool having discontinuous projections on its outer peripheral surface such as to form corresponding radially inward projections on the inner surface of said tube blank.

Preferably, said tool includes at least one tool for fixing said tube blank and at least one gear-like tool, said tool for fixing said tube blank and said gear-like tool are made to roll on the outer surface of said tube blank about the axis of said tube blank thereby to form row or rows of projections on the inner surface of said tube blank.

Embodiments of the invention are further explained by means of drawings, in which

Fig. 1 is a vertical sectional view of a heat transfer tube constructed in accordance with an embodiment
5 of the invention;

Fig. 2 is an enlarged perspective view of an essential part of a heat transfer tube in accordance with the invention;

Figs. 3A, 3B, 3C and 3D are plan views of different
10 embodiments;

Figs. 4A, 4B, 4C and 4D are cross-sectional views of the embodiments shown in Figs. 3A, 3B, 3C and 3D, respectively;

Figs. 5 and 5A are illustrations of an embodiment
15 of the production method in accordance with the invention;

Fig. 6 is an illustration of the operation characteristics of the heat transfer tube in accordance with the invention;

Fig. 7 is a sectional view of a heat transfer tube
20 in accordance with the invention;

1 Fig. 8 is a front elevational view of the heat
exchanger tube;

 Figs. 9 to 11 and Figs. 14 to 17 are illustrations
of experimental data as obtained with heat transfer tubes
5 in accordance with the invention;

 Figs. 12 and 13 and Figs. 18 and 19 are charts
showing the relationship between the pitch of the projections
and the heat transfer rate;

 Figs. 20 and 21 show an example of a heat exchanger
10 tube to which the invention is applied;

 Figs. 22 to 23 are illustration of the performance
of the embodiment shown in Fig. 20; and

 Fig. 24 is an illustration of an example of the use
of the embodiment shown in Fig. 20.

15 An embodiment of the invention will be described
hereinunder with reference to Figs. 1 and 2. This embodi-
ment of the heat transfer tube of the invention has an inner
surface 1 on which are formed projections 3 along a spiral
curve 4. The projection, when viewed in plan, can have a
20 circular form 32 as shown in Fig. 3A, an elliptic form 34 as
shown in Fig. 3B, an asymmetric form 36 as shown in Fig. 3C
or an elongated circular form 38 as shown in Fig. 3D. The
projection has an almost constant cross-sectional shape over
its entire height from the bottom to the top, although the
25 cross-sectional area is progressively decreased from the
bottom towards the top thereof. The vertical section of the

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1 projection also is constituted by smooth curves as shown in
Figs. 4A, 4B, 4C and 4D. The plan shapes as shown in Figs.
3A to 3D are only illustrative and the projection can have
any desired forms resembling those shown in these Figures.

5 A method in accordance with the invention for
producing this heat transfer tube will be explained herein-
under.

Fig. 5 showing an example of the production method
which makes use of a machine having a rotary carrier 50 with
10 a bore for receiving a tube blank and rotatably carrying
three tools 52, 52 and 54 arranged such as to embrace the
tube blank. The tools 52, 52 have smooth outer peripheral
surfaces, while the tool 54 is a gear-like tool having teeth
40 on its surface. As the carrier 50 is driven to rotate
15 around the tube blank by a suitable power, the teeth 40 on
the gear-like tool 54 forcibly depress and plastically
deform the wall of the tube blank thereby forming inward
projections 3 on the inner peripheral surface of the tube
blank. It will be seen that the pitch of the projections 3
20 in the direction of axis 0-0' of the tube blank is determined
by the angle at which the gear-like teeth is mounted. The
configuration of the tooth 40 on the tool 54 is so selected
that the portions of the projection 3 is rounded at corners
thereof corresponding to the corners of the tooth 40.

25 The pitch of the dents on the outer surface of
the tube blank corresponding to the projections 3 is equal
to the circumferential pitch of the teeth 40 on the gear-
like tool 54, while radial the height of the projection 3

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1 can be adjusted by controlling the pressure at which the tool
54 is pressed onto the tube blank. If the tool 54 is driven
in the direction perpendicular to the tube axis, the pro-
jections 3 are formed along independent annular rows.
5 However, if the tube blank 1 is fed axially during the
operation of the tool 54 as shown in Fig. 1, the projec-
tions, 3 are formed along spiral lines. The same effect can
be obtained by feeding the carrier 50 in a spiral manner,
although it is more practicaly to feed the tube in the axial
10 direction while maintaining the carrier 50 stationary.
Smooth surfaces are left between adjacent rows of the
projections. The dents formed in the outer surface of the
tube blank cannot be subjected to the fine machining which
is to be conducted for the purpose of promotion of the
15 boiling and condensation outside the tube, so that only the
smooth areas between adjacent rows of dents are available
as the effective area for promoting the heat transfer. In
order to precisely conduct the required machining on the
tube outer surface, it is necessary that the tube outer
20 surface has areas parallel to the tube axis between adjacent
rows of dents. It will be seen that, the portions of the
tube inner surface under the areas parallel to the tube axis
are naturally formed in parallel with the tube axis.

Fig. 5A schematically shows the gear-like tool
25 used in the described method. It will be seen that the
circumferential pitch z of the projection can be varied by
varying the angle β which is formed between the center of the
tool 54 and the adjacent outer edges of adjacent teeth 40.

1 The tooth height h should be selected to be greater than the
depth of dent from the outer surface of the tube. In a
practical example, the gear-like tool 54 has an outside
diameter D of 33 to 35 mm, a teeth height h of 0.45 to 0.8
5 mm, angle θ of 10 to 20° and a tooth width w of about 1 mm.
Using this gear-like tool, it is possible to obtain a heat
transfer tube having a projection height e of 0.45 to 6 mm
and circumferential projection pitch z of 2.5 to 5 mm.

A change in the outside diameter D naturally
10 requires a change in the angle β . The axial pitch of the
projections can be varied within the range of, for example,
5 to 14 mm, by inclining the gear-like tool 54 at an angle
of 5 to 20° with respect to the tube axis.

Although the embodiment described with reference
15 to Fig. 5 has only one gear-like tool 54 such as to form
the projections 3 along a single spiral curve, the invention
does not exclude the use of a plurality of gear-like tools
54 such that the projections 3 are formed along a plurality
of spiral curves simultaneously. The use of a plurality of
20 gear-like tools 54 is effective in reducing the number of
steps required for the formation of the projection rows,
but this selection depends on the circumferential pitch of
the projections and the axial pitch of the projection rows.

Anyway, according to the described embodiment
25 of the production method of the invention, it is possible
to obtain a heat transfer tube having a plurality of
projections 3 arranged in rows, each projection having a
substantially circularly arched cross-sectional shape and

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- 1 a vertical section constituted by an arcuate protrusion when
taken in a vertical section including the axis of the row of
the projections.

In a particle example, the projection has an
5 elliptic cross-sectional form having a longer diameter rang-
ing between 2 and 5 mm and a shorter diameter ranging between
1.5 and 3 mm.

The rows of the projections may be formed such
that independent conical projections having rounded ends
10 are arranged to protrude from the major level of the tube
inner surface or such that, in each row, the portions bet-
ween adjacent projections are protruded from the major level
of the tube inner surface.

Fig. 6 schematically illustrates the streamlines
15 of a single-phase flow flowing in the tube without making
any phase change. It will be seen that the streamlines 60
in the radially central portion of the tube advances
substantially straight in the direction of the tube axis,
while stream lines 61 near the tube inner surface are
20 deflected by the projections so that vertical eddy currents
having axes in the direction of the tube axis are formed
when these streamlines come out of the spaces between
adjacent projections.

As will be seen from Fig. 7, since the projection
25 3 on the inner surface of the heat transfer tube of the
invention has a smooth and gentle curvature when viewed in
the vertical section, it does not cause any abrupt change
in the directions of the streamlines. Therefore, the

1 effect of the shearing stress due to coherence of the fluid
acting on the tube surface is small and, hence, the pitching
of the tube wall due to the shearing stress can be diminish-
ed advantageously. It is to be pointed out also that, since
5 the cross-section of the projection also has smooth and
gentle configurations, the abrupt deflection of the stream
lines and generation of eddy currents due to exfoliation
are suppressed to minimize the pitching caused by the action
of the fluid.

10 In order to confirm the corrosion resistance of
the heat transfer tube, an accelerated corrosion test was
conducted under the condition shown in Table 1 and results
as shown in Table 2 were obtained.

Table 1 Corrosion Test Conditions

Flow velocity	2 m/sec
Water temperature	40°C
pH	5.0
cl ⁻	600 ppm
Testing period	30 days

Table 2 Results of Corrosion Test

Shape of projection	Corrosion rate (mm/year)
Two-dimensional (continuous projection)	0.56
Three-dimensional (angular projection)	0.77
Three-dimensional (rounded projection)	0.54

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1 From Table 2, it will be seen that rounded projec-
tions can retard the corrosion as compared with angular
three-dimensional projections and can provide a corrosion
rate which is as small as that observed with heat transfer
5 tubes having two-dimensional projections which are known as
exhibiting excellent corrosion resistance. Thus, the
corrosion rate in the heat transfer tube with rounded three-
dimensional projections shown in Table 2 is practically
acceptable.

10 An explanation will be made hereinunder as to the
performance of a heat transfer tube of the invention having
rounded projections. An experiment was conducted by varying,
among the parameters which affect the performance of the heat
transfer tube, the projection height, circumferential pitch
15 of projections and the axial pitch of the projection, in
order to confirm the effect of the invention. The heat
transfer tube subjected to the experiment has an inside
diameter d which ranges between 14.7 mm and 15.8 mm.

Fig. 9 shows the values of heat transfer rate and
20 the pressure drop as obtained when the projection height e
was 0.45 mm (marked at Δ), 0.5 mm (marked at Δ) and
0.6 mm (marked at \square), while the axial pitch P and the
circumferential pitch z were fixed at 7 mm and 4 mm, respec-
tively. In Fig. 9, the axis of abscissa represents Reynold
25 number and the drag coefficient f which represents the
coefficient of flow resistance along the tube. As is well
known, the reynolds number Re is given by the following
formula:

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$$R = u \cdot d / \nu$$

1 where, u represents the mean flow velocity of the fluid in the tube (m/s), d represents the inside diameter of the tube (mm), and ν represents the kinematic coefficient of viscosity of the fluid (m^2/s).

5 The axis of ordinate shows dimensionless heat transfer rate $Nu/Pr^{0.4}$ which is given by the following formula:

$$Nu/Pr^{0.4} = \alpha d / \lambda / Pr^{0.4}$$

where, α represents the heat transfer coefficient ($W/m^2 K$), λ represents the heat conductivity of the fluid ($W/m K$) and
10 Pr represents the Prandtl number of the fluid.

A comparison test was conducted using a comparison tube having a smooth inner surface which has not been subjected to any machining. This comparison tube showed heat transfer rate which well approximates the value given
15 by $Nu = 0.023 Re^{0.8} Pr^{0.4}$ (shown by curve A) which is known as "Dittus-Boelter" formula. The comparison tube showed also a drag coefficient which well approximates the value given by $1/\sqrt{f} = 2.0 \log (Re \sqrt{f}) - 0.8$ (curve B) which is known as "Prandl's equation". For the purpose of
20 clarification of the drawing, the heat transfer rate and drag coefficient as obtained with the comparison tube are not shown in Fig. 9. The comparison tube had an inside diameter of 15.8 mm. It will be seen that the samples heat transfer tube of the invention having projection heights of

1 0.5 mm and 0.6 mm showed performance which is about twice
as high as that of the comparison tube having smooth inner
surface.

From Fig. 9, it will be seen also that the drag
5 coefficient is increased at a rate greater than the rate of
increase of the heat transfer coefficient as the projection
height e is increased. Therefore, when the projection
height e is increased above a predetermined threshold, the
effect of the increase in the heat transfer rate is exceeded
10 by the loss caused by the pressure drop. More specifically,
in the case of the arrangement shown in Fig. 9, when the
projection height is increased above 0.5 mm, the effect of
promotion of heat transfer is reduced because of a large
increase in the drag coefficient in contrast to a small
15 increase in the heat transfer rate. From this fact, it is
understood that the projection height is optimumly 0.5 mm,
in the case of the heat transfer tube explained in connection
with Fig. 9.

In order to confirm the above-explained advantage-
20 ous effect of the invention, a reference shall be made to
a literature which has been authorized as a reference
concerning the heat transfer rate and the drag coefficient.

An example of such literatures is "Application of
Rough Surfaces to Heat Exchanger Design" by R.L. Webb and
25 E.R.G. Eckert, International Journal of Heat and Mass
Transfer, Vol. 15, p 1647-1658, 1972. In this literature,
there is given a concept concerning the heat transfer rate
and the drag resistance as expressed by:

$$\frac{(St/St_0)}{(f/f_0)^{1/3}}$$

1 where, the suffix 0 (zero) represents the values as obtained
with tube having smooth inner surface.

An evaluation was conducted by computing the ratios
appearing in the above formula. In case of the tube having
5 a smooth inner surface, the values of the ratios are "1".
The value given by the formula is increased as the heat
transfer performance is improved. The experimental data
shown in Fig. 9 were obtained with the water flow velocity
of 2.5 m/sec and Reynolds number Re of 3×10^4 which is
10 calculated from the physical values corresponding to the
water temperature in the refrigerator to which the heat
transfer tube of the invention is applied. Fig. 10 shows
the value of the above-mentioned formula in relation to the
projection height.

15 From Fig. 10, it will be seen that the best
performance is obtained when the projection height is around
0.5 mm, and the performance is progressively degraded as
the projection height is increased beyond 0.5 mm and reduced
below 0.5 mm. The optimum projection height is related to
20 the boundary layer of the fluid adjacent the tube surface
and can be considered as being almost constant, although
there may be a small difference by factors such as, for
example, the tube diameter. In Fig. 10, a symbol D pointing
a value 1.43 represents the value of the formula mentioned
25 above, calculated for a known heat transfer tube having two-

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- 1 dimensional ribs ($e = 0.3$ mm, $P = 4$ mm) of the type shown
in United States Patent No. 3,768,291. Thus, the performance
of the heat transfer tube of the invention having three-
dimensional projections exceeds the level of $D = 1.43$
5 exhibited by the known heat transfer tube, when the pro-
jection height ranges between about 0.45 mm and 0.6 mm.

A description will be made hereinunder as to the
result of an experiment which was conducted by using models
in order to examine the influence of the circumferential
10 pitch z of the projection on the heat transfer performance.

Fig. 11 shows the heat transfer rate and the drag
coefficient as measured with three different values of the
circumferential pitch z of the projections ($z = 2.5$ mm
marked at Δ , $z = 4$ mm marked at \circ and $z = 5$ mm marked at
15 \square), while fixing the axial pitch of the projection and the
projection height at 7 mm and 0.45 mm, respectively. From
Fig. 10, it will be seen that a higher heat transfer rate
is obtained when the circumferential pitch z is 4 mm than
when the same is 2.5 mm. It will be seen also that the
20 drag coefficient f is greater when the circumferential
pitch z is 2.5 mm than when the same is 4 mm. These facts
tell that a higher heat transfer performance is obtained
when the circumferential pitch z is 4 mm than when the same
is 2.5 mm.

25 When the circumferential pitch z is 2.5 mm,
adjacent projections 5 and 5 are substantially connected to
each other so that there is no clearance C between adjacent
projections, as will be seen from Fig. 12. Therefore, in

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1 this case, the size of vertical eddy currents 6 (see Fig.
13) produced by the stream lines coming out of the space
between adjacent projections is small as represented by 7.
Thus, the smaller circumferential pitch z makes the
5 haracteristics of the three-dimensional projections approach
those of the two-dimensional projections, so that the heat
transfer performance becomes closer to that of the heat
transfer tube having two-dimensional projections. In Fig.
11, the curve plotted along the values marked by \diamond measured
10 with a heat transfer tube having two-dimensional projections
($p = 7$ mm, $e = 0.5$ mm), together with the values measured
with the heat transfer tube having three-dimensional projec-
tions. It will be seen that Fig. 11 also shows the same
tendency, i.e., the fact that the smaller circumferential
15 pitch z causes an increase in the pressure drop such as to
approximate that provided by the two-dimensional projections.

When the circumferential pitch z is 4.5 mm,
vertical eddy currents 6 having rotation axes parallel to
the flowing direction are emitted from the clearances C
20 between adjacent projections such as to enhance the heat
transfer. In the case of two-dimensional projections, the
streamlines are exfoliated when they pass over the two-
dimensional projections and re-attach to the tube surface in
the area downstream from the projections, and the promotion
25 of the heat transfer owes to this re-attaching of the
streamlines. In contrast, in case of the three-dimensional
projections, the promotion of heat transfer owes to the
generation of vertical eddy currents, so that the energy

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1 of the stream can be utilized more efficiently than in the
case of the two-dimensional projections. In this case, the
clearance c between adjacent projections was 1 mm, while the
length b of each projection was 3 mm. When the clearance
5 c is increased to a certain amount, the vertical eddy cur-
rents which are effective in the promotion of heat transfer
are not produced so that the heat transfer promotion effect
is not so high. Referring to Fig. 11, when the circum-
ferential pitch z is 5 mm (see marke □), the increment of
10 the heat transfer rate is smaller than that obtained when
the pitch z is 4 mm. This suggests that the increase of the
clearance c reduces the heat transfer rate.

In this case also, the values obtained through
the test were evaluated by making use of the aforementioned
15 formula $St/St_0/(f/f_0)^{1/3}$. The result is shown in Fig. 14
from which it will be seen that the highest heat transfer
performance is obtained when the circumferential pitch z is
4 mm. The value denoted by D was obtained with the two
dimensional rib ($e = 0.3$ mm, $p = 4$ mm). The value D
20 suggests that the three dimensional projections provide
higher heat transfer promotion effect. More specifically,
the three dimensional projections provide the higher effect
than that calculated from the values obtained through
experiment with the heat transfer tube having two-dimensional
25 ribs when the circumferential pitch z ranges between 3.5 mm
and 5 mm and, therefore, this range is selected as being
the preferred range of the circumferential pitch.

In order to examine the influence of the axial

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1 pitch, experiment was conducted by using three different
values of axial pitch: namely, 5 mm, 7 mm and 10 mm, while
fixing the rib height e and the circumferential pitch z at
0.5 mm and 4 mm, respectively. The result of this experiment
5 is shown in Fig. 15. More specifically, Fig. 15 shows the
heat transfer rate and the drag coefficient as obtained when
the axial pitch is 5 mm (mark ∇), 7 mm (mark Δ) and 10 mm
(mark \square). It will be seen that both the heat transfer
rate and the drag coefficient are increased as the axial
10 pitch is increased. As in the preceding cases, the values
obtained in this experiment were evaluated by using the
formula $(St/St_0)/(f/f_0)^{1/3}$, the result of which is shown in
Fig. 16. From this Figure, it will be seen that the axial
pitch of 5 mm and 7 mm provide substantially equal values
15 of the ratio given by the above-mentioned formula, while the
axial pitch of 10 mm provides a considerably smaller value.
This is attributable to the following reason. Referring to
Figs. 17a and 17b, the promotion of heat transfer owes to
the eddy currents generated by the three-dimensional projec-
20 tion 3, so that the high heat transfer performance is
maintained when the next projection exists within the length
in which the eddy currents are diffused and extinguished,
as shown in Fig. 17a. The length in which the eddy currents
are extinguished is about 10 times as large as the projection
25 height, when the projection is two-dimensional. Namely,
when the projection height is 0.5 mm, the length l is given
as $0.5 \text{ mm} \times 10 = 5 \text{ mm}$. Thus, the length l shown in Fig. 17a
is estimated to be about 5 mm. Thus, the high performance is

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1 obtained when the axial pitch is between 5 and 7 mm. However, when the axial pitch is 10 mm, the pitch p is greater than the length l as shown in Fig. 17b. In this case, the eddy currents are extinguished before reaching the next
5 projection so that there exists a large area where there is no eddy current, resulting in a smaller heat transfer promotion effect. In Fig. 16, D represents the value which is calculated in accordance with the aforementioned formula $(St/St_0)/(f/f_0)^{1/3}$ from the values obtained through an
10 experiment with the heat transfer tube having two-dimensional ribs. The axial pitch is preferably selected to range between 5 mm and 9 mm because this range provides both the heat transfer performance higher than the value D and easy fabrication of the heat transfer tube.

15 Preferred sizes of the projections have been discussed on the basis of experimental data, and it has been confirmed that the projection height, circumferential pitch of projection and the axial pitch of the projection preferably range between 0.45 and 6 mm, 3.5 and 5 mm and 5 and
20 9 mm, respectively, in order to attain an appreciable effect in the improvement in the heat transfer performance.

The pattern of streamlines past the rows of rounded projection varies depending on the arrangement of the projections. For instance, Fig. 18 shows the case where
25 the projections 3 are arranged in a staggered manner. In this case, the heat transfer promotion effect is obtained by the fact that the streamlines 90 after passing the clearance between adjacent projections collide with the

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1 projection on the downstream side. However, when the
projections 3 are arranged regularly in a lattice-like form
as shown in Fig. 19, the vortex flow in the streamline 100
downstream from the projection 3 collides with the down-
5 stream projection before the energy of the vortex flow is
diffused, so that the heat transfer promotion effect is
suppressed. In addition, the streamlines which have passed
through the clearance between adjacent projections are
straight and parallel to the tube axis so that it does never
10 contributes to the heat transfer promotion effect. For this
reason, the projections are preferably arranged in a stagger-
ed manner.

In the case of conventional heat transfer tube
with continuously corrugated inner surface, i.e., heat
15 transfer tube with two-dimensional ribs, the pressure drop
is considerably high although the heat transfer performance
is excellent as shown in Fig. 11. The pressure drop is
preferably small because the large pressure drop requires
a greater pumping power for circulating the liquid. In case
20 of the heat transfer tube of the invention, the increment
in the heat transfer rate allows a reduction in the heat
transfer area for a given thermal load, so that the pressure
drop is decreased correspondingly such as to compensate for
any reduction of the performance due to the increase in the
25 drag coefficient.

Since the generation of turbulent flow in the area
adjacent the tube wall is not so much affected by the tube
diameter, the heat transfer tube of the invention having

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- 1 three-dimensional projections can be applied to tubes having inside diameters of about 10 to 25.4 mm.

Obviously, the heat transfer tube of the invention can have a suitable construction for promoting the heat transfer also on the outer surface thereof. The heat transfer promoting construction on the outer surface can be formed, for example, by the following procedure.

As the first step, projections are formed on the inner surface of the tube blank by means of rolls which act on the outer side of the tube blank. The dents in the outer surface of the tube blank, which have been formed by the rolls for forming the projections on the inner surface of the tube blank, cannot be machined finely for the purpose of improving the heat transfer. It is, therefore, necessary to form the heat transfer promoting construction on the portions of the tube outer surface which are parallel to the tube axis and devoid of the dents. Therefore, in the next step of the process, porous heat transfer surfaces 208 which effectively promote the boiling heat transfer are formed in the smooth areas 207 of the tube outer surface devoid of the dents which have been formed during the forming of the projections on the inner surface, as shown in Fig. 20. In this Figure, a reference numeral 230 denote the dents formed when the projections on the inner surface were formed.

25 The fine machining on the outer surface of the tube block for the promotion of heat transfer may be conducted before the formation of the projections on the inner surfaces. In such a case, however, the heat transfer

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1 promoting construction formed by the fine machining tends
to be collapsed by the rolls which act on the outer surface
during the forming of the projections on the inner surface.
Therefore, in the described case, the fine machining on the
5 outer surface is conducted after the formation of the pro-
jections on the inner surface.

The fine machining on the outer surface of the tube
blank is conducted, for example, in the following way. As
the first step, shallow grooves of 0.1 to 0.2 mm are formed
10 at an angle of about 45° to the tube axis by knurling. Then,
the knurled surface is ploughed by a cutting tool substan-
tially perpendicularly to the tube axis such as to form
fins 212. The height and the pitch of the fins 212 are
preferably about 1 mm and 0.4 to 0.6 mm, respectively. In
15 consequence, rows of saw-teeth-shaped fins are formed on
the smooth areas of the tube blank. Subsequently, the fins
are made to laid down or collapsed such that adjacent fins
get closer to each other by, for example, knurling, thereby
forming a porous construction 208 constituted by fine
20 cavities 209 which open to the outside through fine open-
ings 210 between adjacent fins, as shown in Fig. 20. The
thus formed tube has an outer surface as shown in Fig. 21.

In the use of this heat transfer tube, water is
circulated through the tube while freon gas which is an
25 organic medium having a low boiling point flows outside the
tube. The tube is most probably used in a shell-and-tube
type heat exchanger having a multiplicity of such tubes
arranged in a barrel and used as, for example, as an

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1 evaporator of a turbo-refrigerator. In such a case, the
temperature of the water inside the tube is usually about 5
to 10°C higher than the freon outside the tube. The flow
of water in the tube has turbulency which is produced in
5 the area near the tube inner surface due to the presence
of the projections, so that the heat exchange between the
tube inner surface and the water is made more effectively
than in the case where the tube inner surface is smooth.

On the other hand, the freon flowing outside the
10 tube is boiled to produce voids. These voids, once generat-
ed, are trapped in the cavities such as to form this freon
films between the surfaces of the cavities and the voids.
This thin freon film is easily evaporated such as to pro-
mote the heat transfer by the phenomenon called latent heat
15 transportation.

Fig. 22 shows the influence of the pitch p of
the projections in the heat transfer tube shown in Fig. 21,
on an assumption that the projection height is 0.3 mm. As
will be seen from this Figure, there is a certain range of
20 pitch p which provides high heat transfer efficiency.
Namely, when the pitch p is large, the tube has a large
smooth area on the outer surface thereof, so that the
porous heat transfer promoting construction can be formed
over a wide area. In consequence, the heat transfer between
25 the outer tube surface and the medium flowing outside the
tube is increased correspondingly.

On the other hand, an increased pitch p on the
tube inner surface increases the area where the turbulency

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1 70 of the streamline caused by the projection 3 does not
affect the region near the inner tube surface. In con-
sequence, the heat transfer rate is drastically decreased.
In this case, the reduction in the heat transfer performance
5 by the forced convection in the inner side of the tube
exceeds the increment of the heat transfer performance
obtained at the outer side of the tube. Consequently, the
overall heat transfer performance of the tube as a whole
is drastically decreased as the pitch p is increased beyond
10 a certain value. On the other hand, the increase of the area
on the tube inner surface on which the heat transfer is
improved by the turbulency is saturated when the pitch p
is reduced below a certain value, so that no substantial
increase in the heat transfer efficiency by the forced
15 convection inside the tube is attained. On the other hand,
the smaller pitch p of the projections causes a drastic
reduction in the area having the heat transfer promoting
construction on the tube outer surface so that the boiling
heat transfer on the outer tube surface is decreased
20 drastically. Consequently, the overall heat transfer rate
is decreased when the pitch p is decreased below a certain
value.

For these reasons, high overall heat transfer rate
of the heat transfer tube can be obtained only when the pro-
25 jection pitch p falls within a predetermined range. In the
case of the arrangement shown in Fig. 22, the optimum range
is between 5 mm and 15 mm.

The heat transfer tube of the invention can be used

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1 in a shell-and-tube type heat exchanger. In such a case,
the heat-exchanger is produced by expanding the tube at its
both ends 215 as shown in Fig. 24, forming the projections,
inserting the tube into corresponding holes in end plates
5 216 and then fixing the tube to these end plates by expand-
ing the tube ends. The conventional method of forming
projections by means of the plug or by drawing cannot be
conducted unless both ends of the tube are left straight.
Therefore, when these conventional methods are used, the
10 projections are first formed on the tube inner surface and
then the projections on both ends of the tube are removed
by cutting such as to smooth the surfaces at both ends of
the tube, before the tube ends are expanded. Thus, the heat
transfer tube of the invention is advantageous also in that
15 it can reduce the number of steps in the assembly of a shell-
and-tube type heat exchanger.

Claims:

1. A heat transfer tube for single-phase flow having one or more rows of projections (3) formed on the inner surface (1) of said tube along a spiral curve or curves (4), each row having a multiplicity of projections (3) formed discontinuously, the portions of the tube inner surface (1) between adjacent rows presenting surfaces parallel to the tube axis, c h a r a c t e r i z e d in that each of said projection (3) has a circular (32) or elliptic (34) or a similar cross-section (36, 38) constituted by smooth curves at its any portion along the height including the bottom thereof, the cross-sectional area of said projection progressively decreases towards the top thereof.
2. A heat transfer tube for single-phase flow according to claim 1, wherein said projections have a height of 0,45 to 0,6 mm and are arranged at a circumferential pitch of 3,5 to 5 mm.
3. A heat transfer tube for single-phase flow according to claim 1 or 2, c h a r a c t e r i z e d in that said projections (3) are arranged at an axial pitch of 5 to 15 mm.
4. A heat transfer tube for single-phase flow according to one of the claims 1 to 3, c h a r a c t e r i z e d by porous heat transfer surfaces (208) formed on the outer surface (207) of said tube.
5. A method of producing a heat transfer tube for single-phase flow having one or more rows of projections (3) formed by a plastic work on the tube inner surface (1) along a spiral curve or curves (4), each row including a multiplicity of discontinuous projections (3) such that the portions of the tube inner surface (1)

between adjacent rows presenting surfaces parallel to the tube axis, c h a r a c t e r i z e d by: depressing the outer surface of a tube blank by a tool
10 (54) having discontinuous projections (40) on its outer peripheral surface such as to form corresponding radially inward projections (3) on the inner surface of said tube blank.

6. A method of producing a heat transfer tube for single-phase flow according to claim 5, c h a r a c t e r - i z e d in that said tool includes at least one tool (52) for fixing said tube blank and at least one gear-
5 like tool (54), said tool (52) for fixing said tube blank and said gear-like tool (54) are made to roll on the outer surface of said tube blank about the axis of said tube blank thereby to form row or rows of projections (3) on the inner surface of said tube blank.

FIG. 1

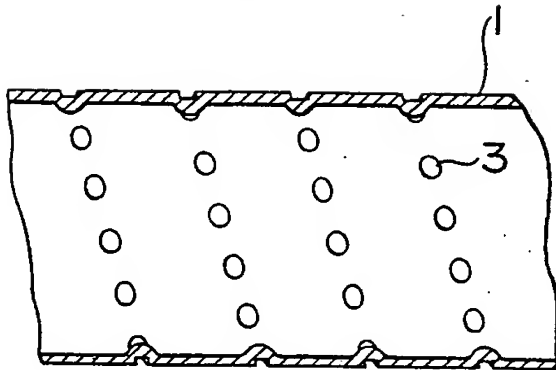


FIG. 2

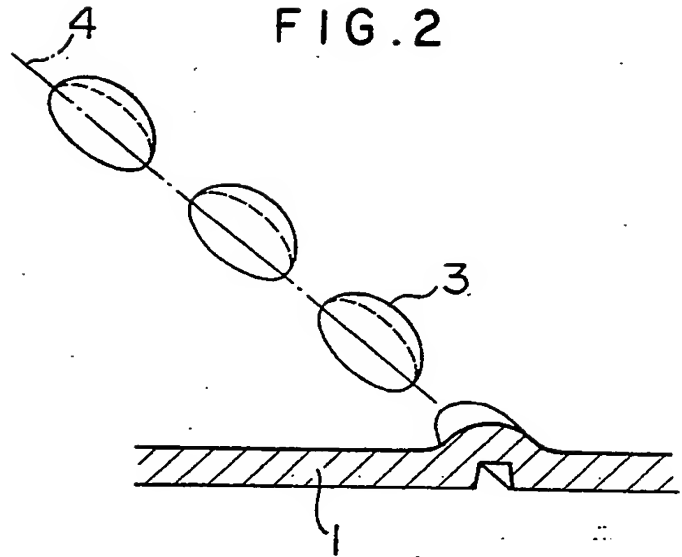


FIG. 3A

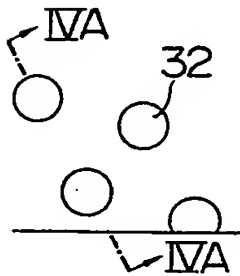


FIG. 3B

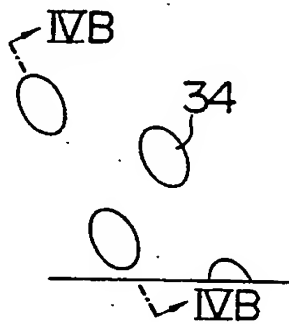


FIG. 3C

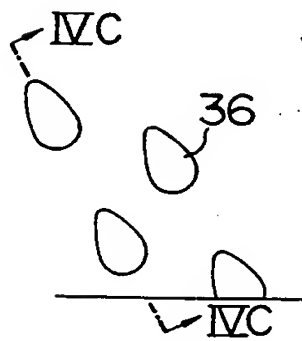


FIG. 3D

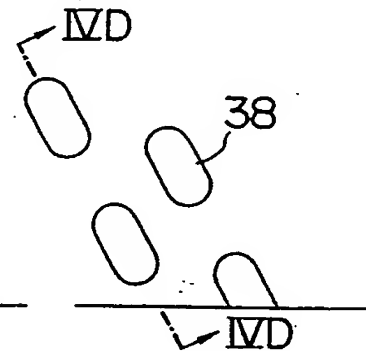


FIG. 4A

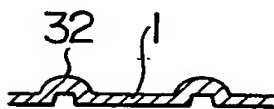


FIG. 4B

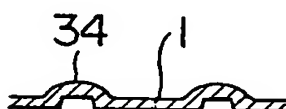


FIG. 4C

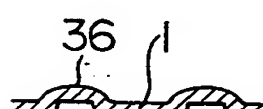


FIG. 4D

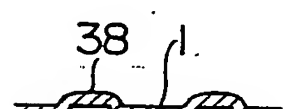


FIG. 5

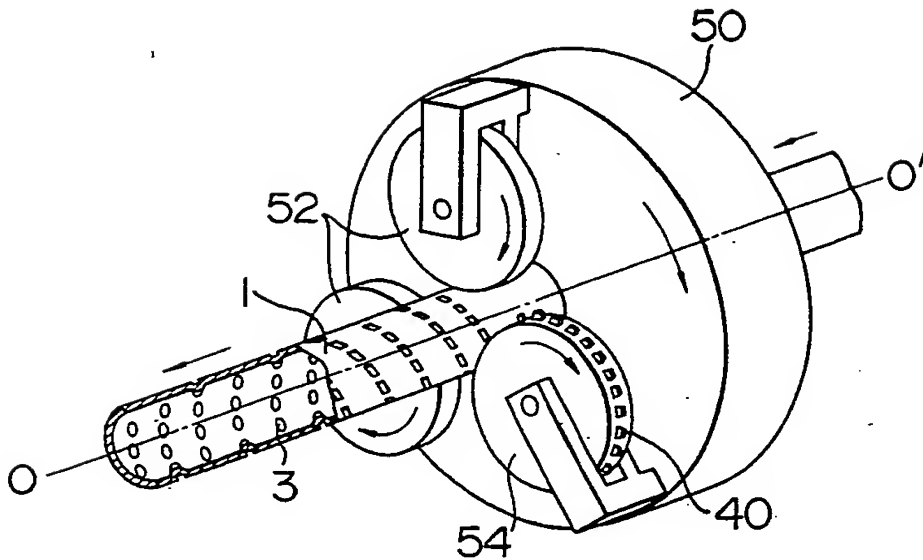


FIG. 5A

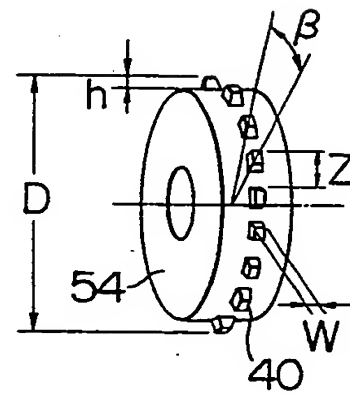


FIG. 6

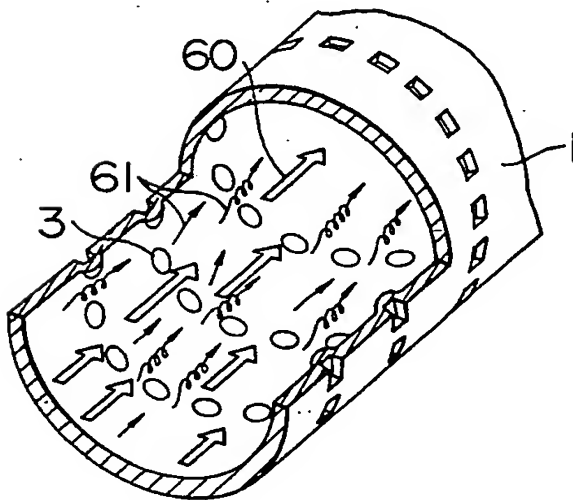


FIG. 7

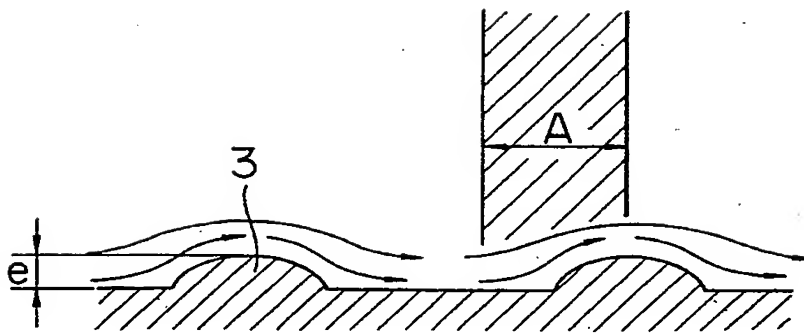


FIG. 8

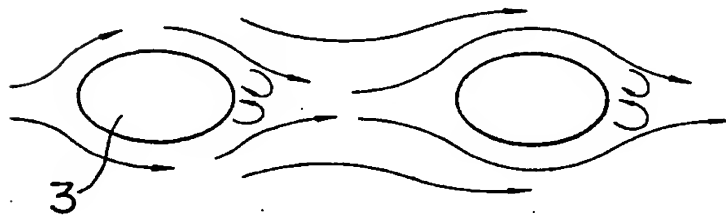


FIG. 9

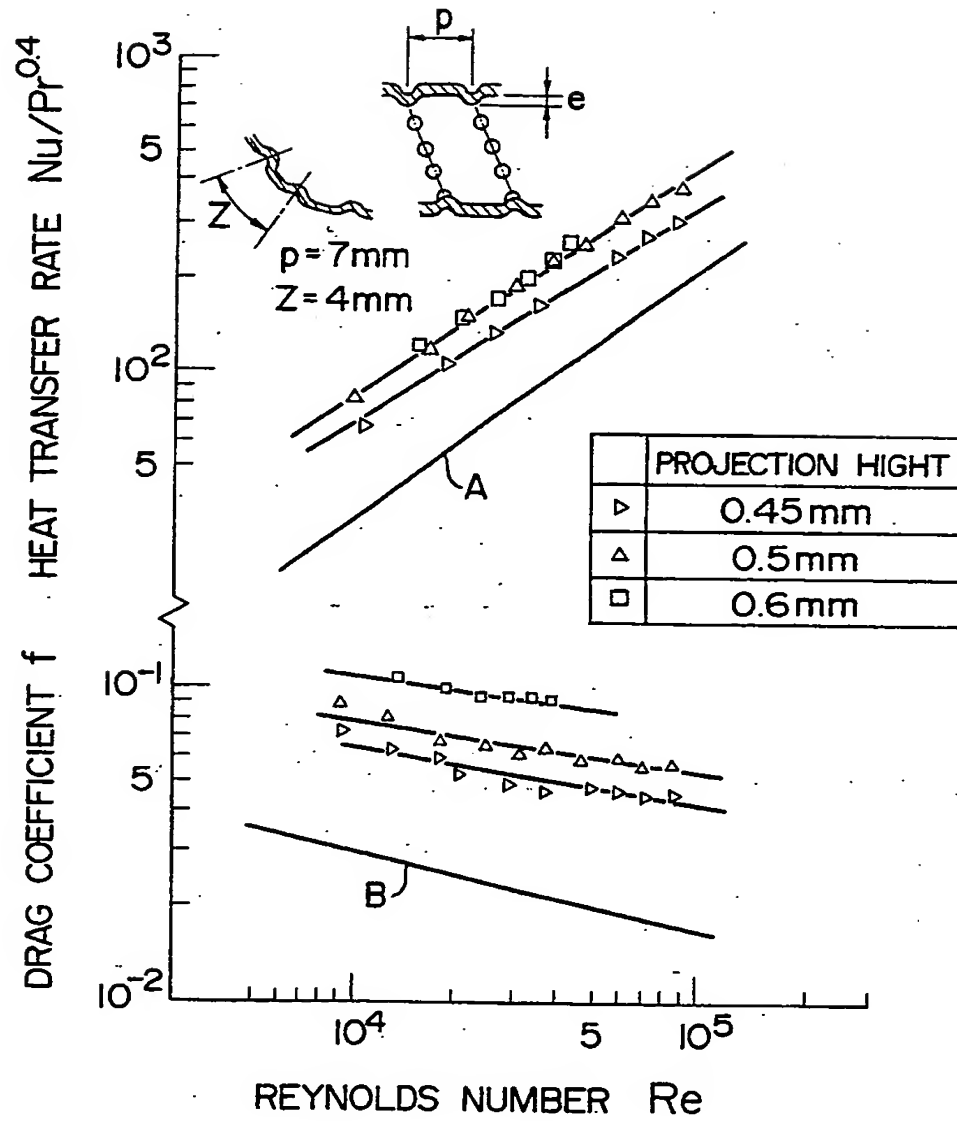


FIG. 10

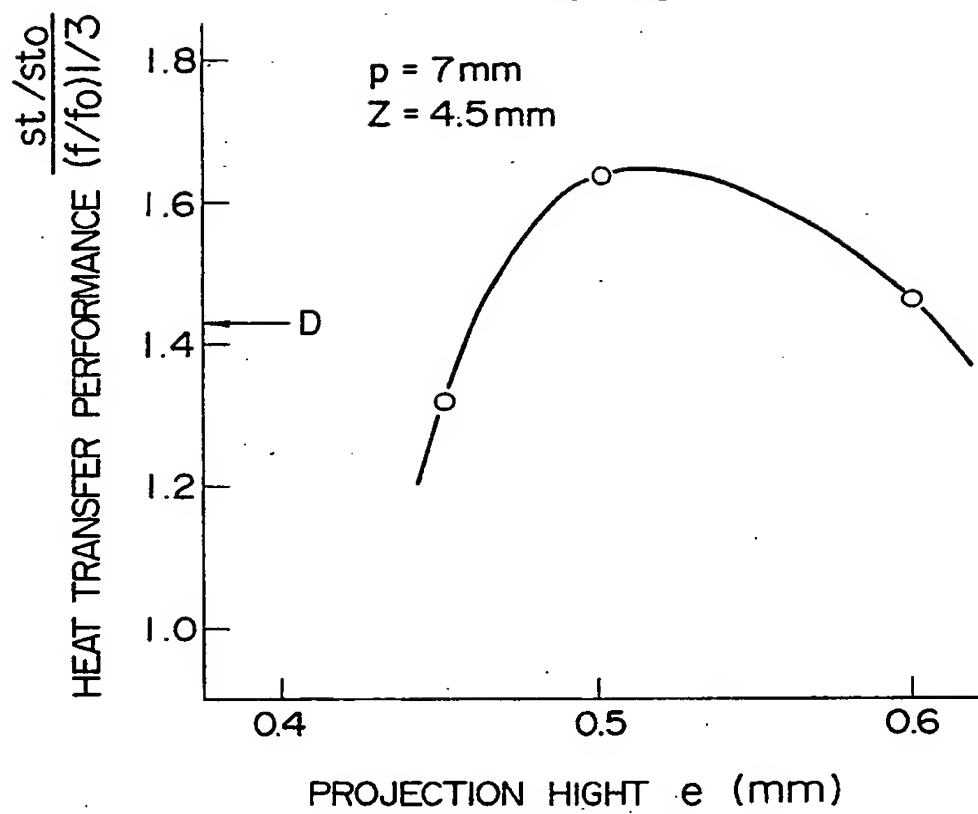


FIG. II

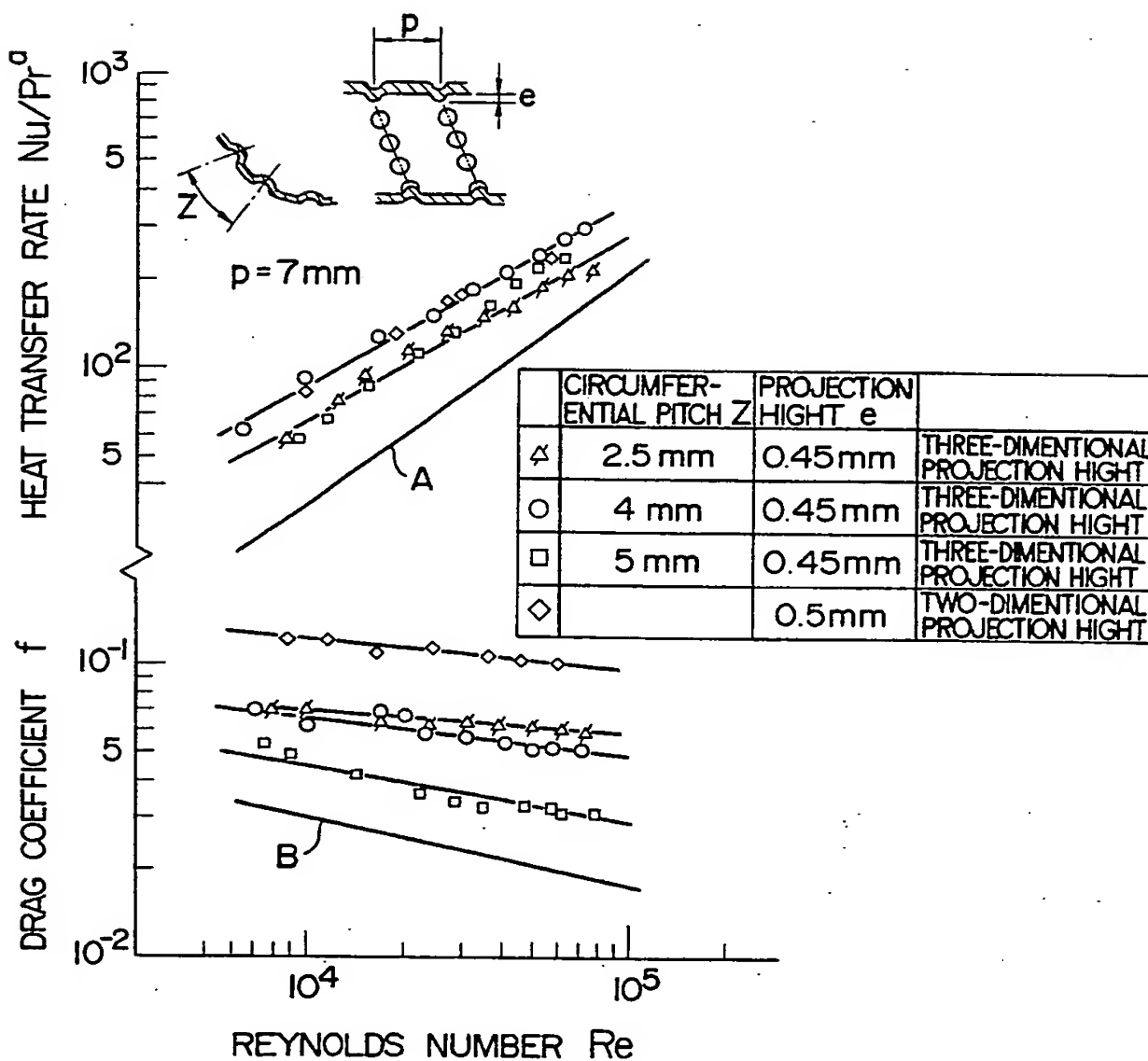


FIG. 12

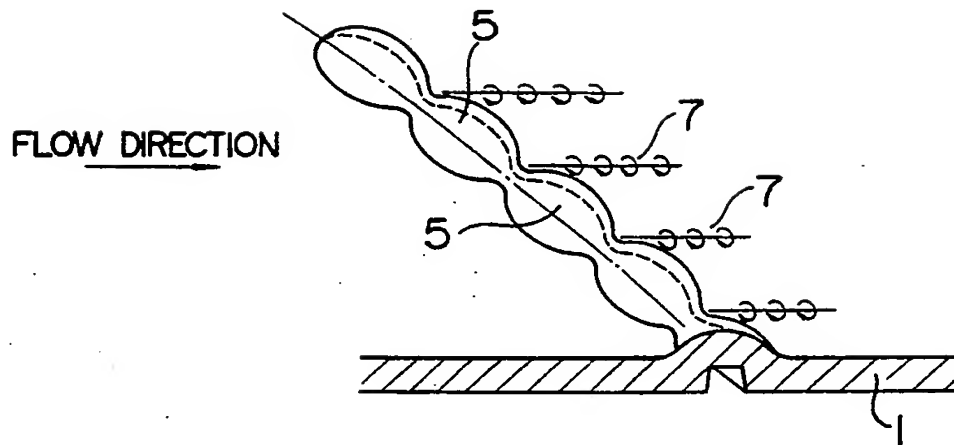


FIG. 13

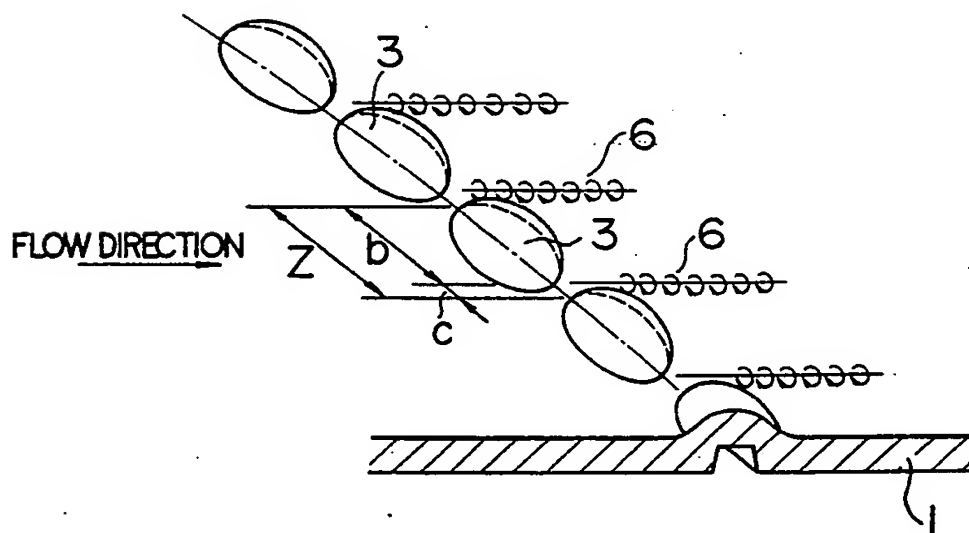


FIG. 14

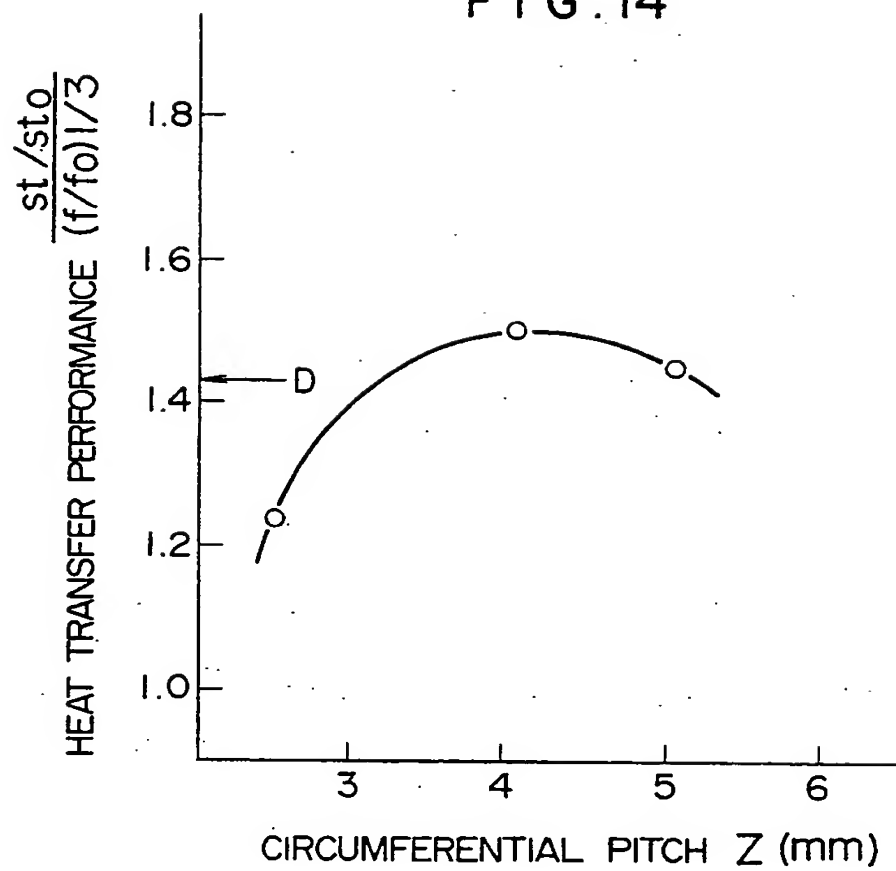


FIG. 15

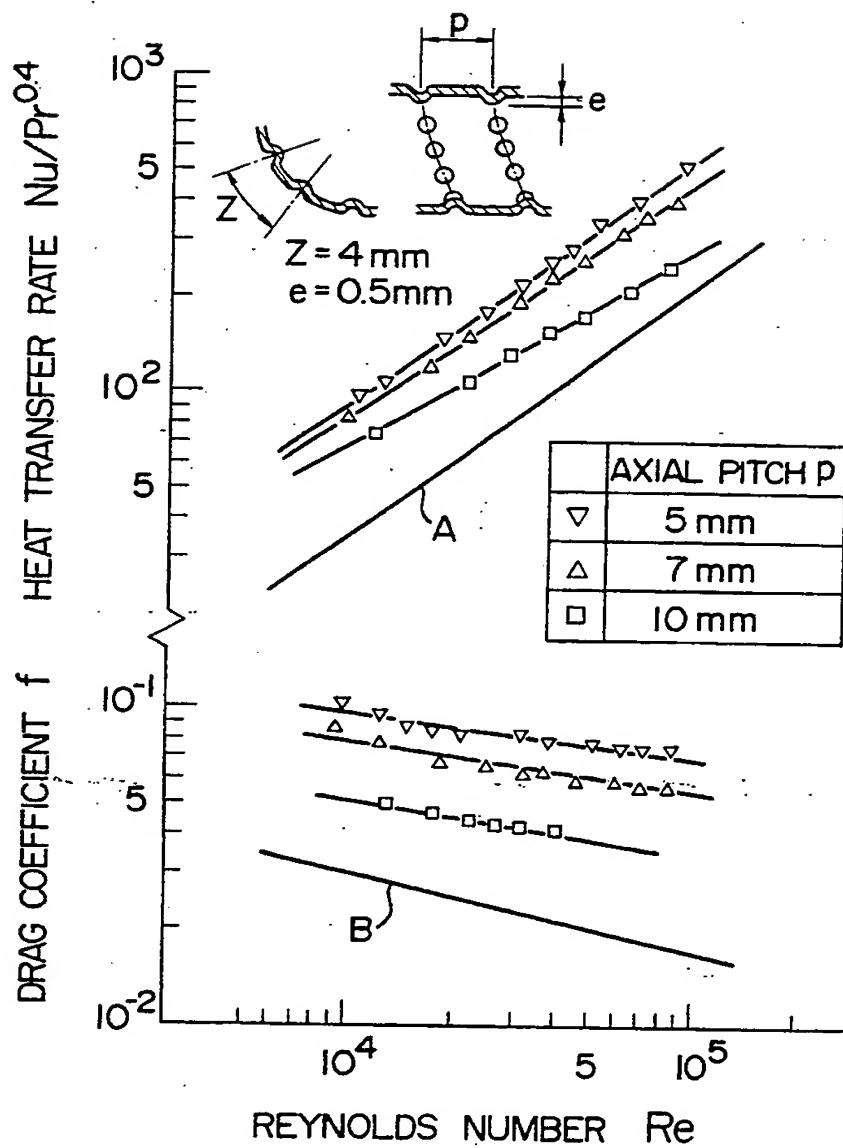


FIG. 16

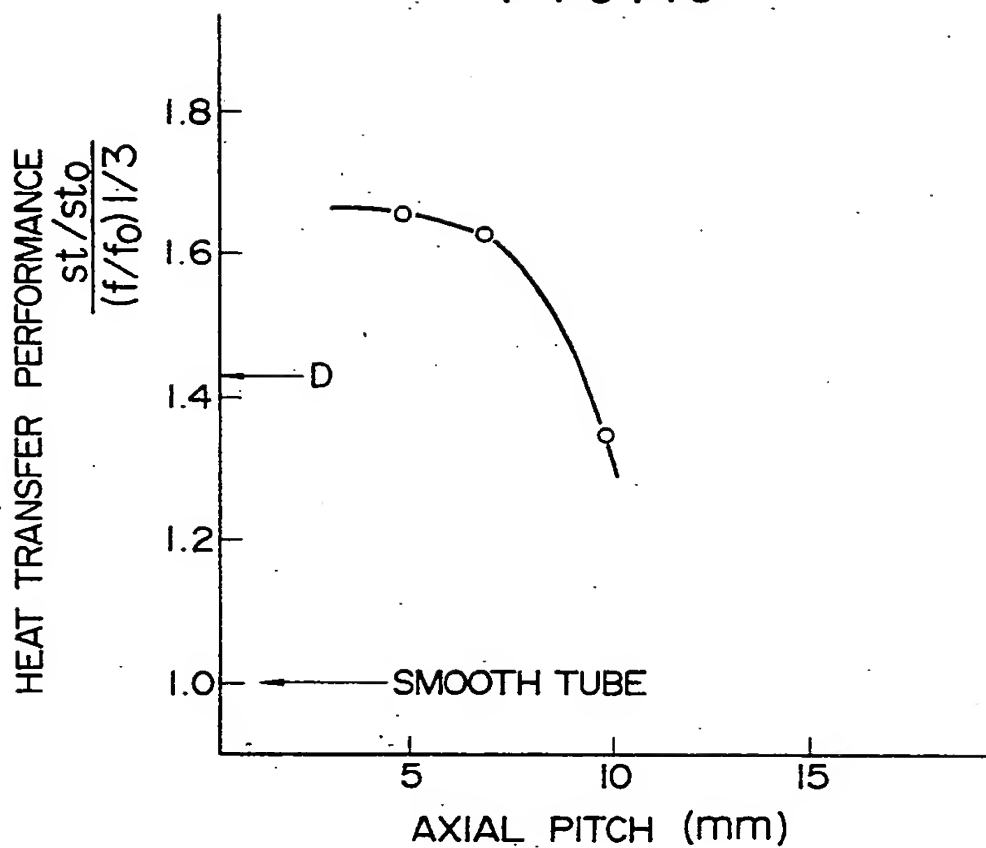


FIG. 17a

FLOW DIRECTION

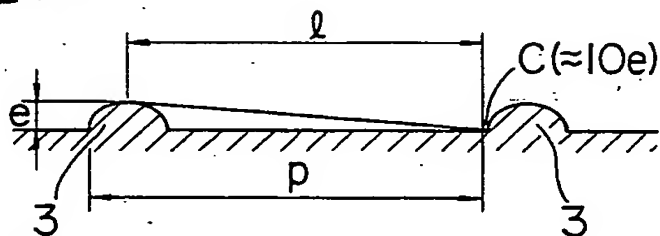


FIG. 17b

FLOW DIRECTION

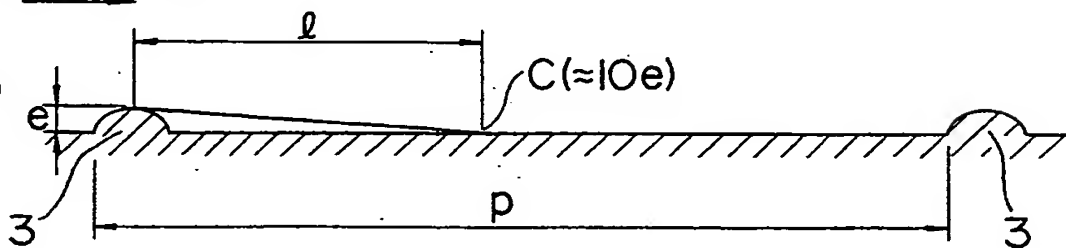


FIG. 18

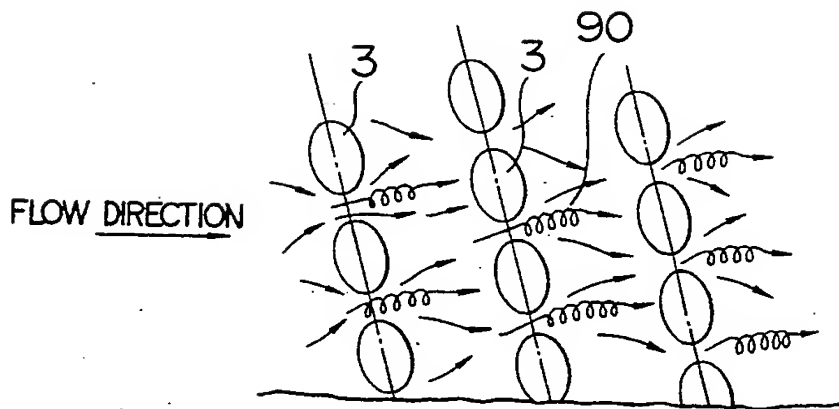


FIG. 19

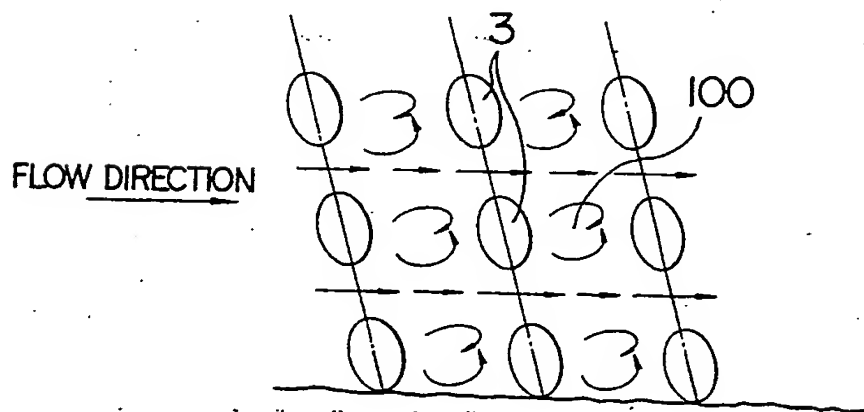


FIG. 20

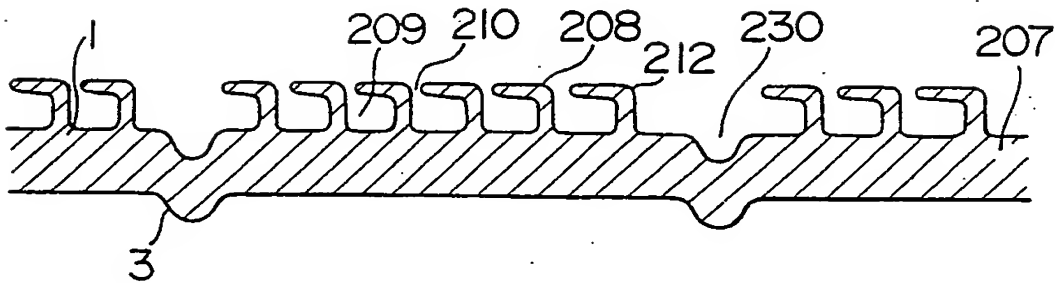


FIG. 21

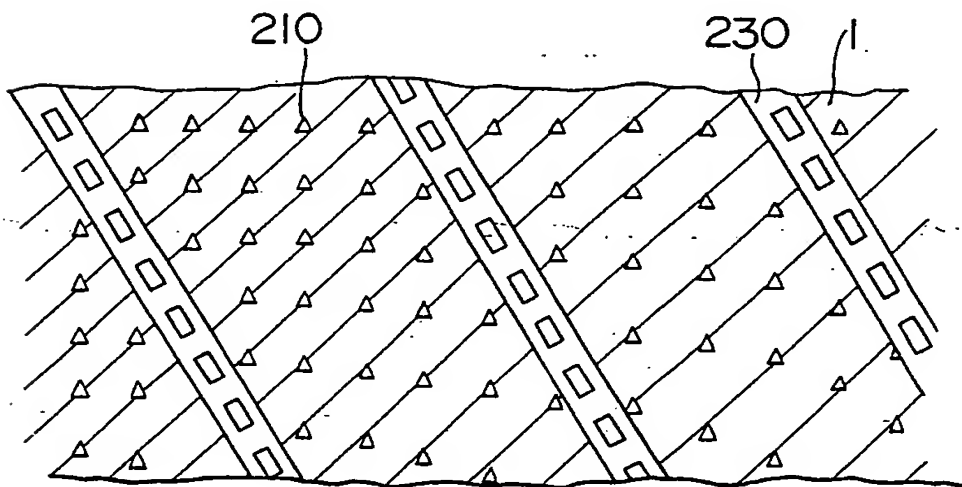


FIG. 22

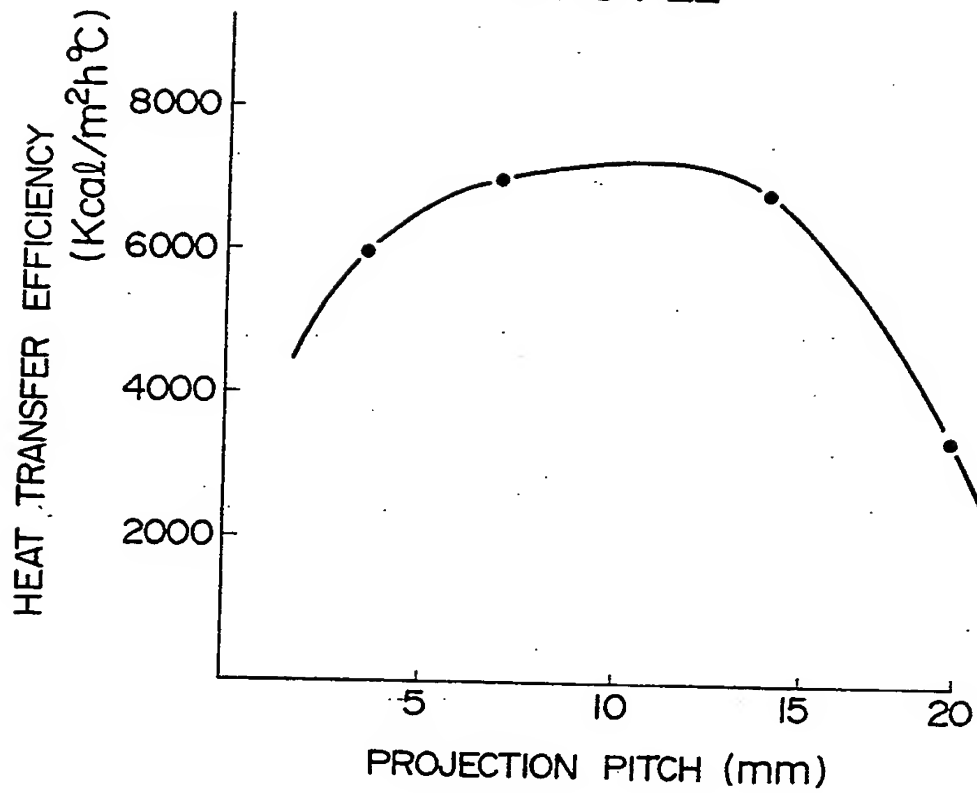


FIG. 23

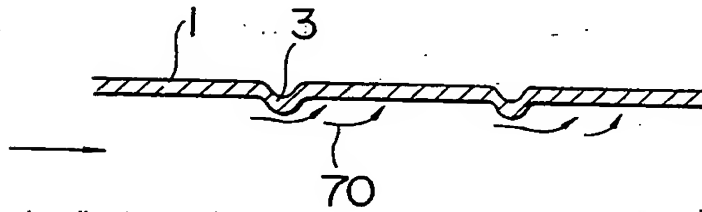


FIG. 24

